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First Named Inventor	:	Huaiyu PAN		
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Title	:	Pulse Tube Refrigerator		

SUBMISSION OF PRIOR ART DOCUMENTS FOR THE FILE

Mail Stop IF **May 20, 2005**
Commissioner for Patents
P.O. Box 1450
Alexandria, VA 22313-1450

Sir:

Attached hereto are copies of the following two documents:

Radebaugh, "Advances in Cryocoolers", ICEC 16/1CMC Proceedings; and

Wang et al, "0.5W CLASS TWO-STAGE 4 K PULSE TUBE CRYOREFRIGERATOR", Advances in Cryogenic Engineering, Volume 45, Plenum Publishers, 2000.

Applicants have reviewed these documents, and do not consider that they are "material" in the sense of Rule 56, in that they disclose only background information concerning the existence of 4 K Pulse Tube Refrigerators, and the difficulties encountered in optimizing the performance of Pulse Tube Refrigerators. Accordingly, they are being submitted to the Patent and

Trademark Office for the record. Applicants accordingly request that these documents be placed in the file of the present application.

If there are any questions regarding this response or the application in general, a telephone call to the undersigned would be appreciated since this should expedite the prosecution of the application for all concerned.

Respectfully submitted,

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Advances in Cryocoolers

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Cryocooler problems such as poor reliability and high cost have limited the application areas for cryogenics. Recent advances in cryocoolers are beginning to alleviate some of these problems. Applications for such cryocoolers are presented and the requirements which these applications impose on the cryocoolers are reviewed. Most applications require improved reliability. For space applications the additional requirement of high efficiency is needed, whereas for commercial applications low cost is also a primary factor. Developments within the last 10 years to meet some of these requirements include the use of flexure and gas bearings to eliminate rubbing contact, pulse tube refrigerators to eliminate the moving displacer in Stirling and Gifford-McMahon refrigerators, the use of gas mixtures to improve efficiency of Joule-Thomson refrigerators and to allow the use of commercial compressors, the use of sorption compressors and thermoacoustic drivers to replace mechanical compressors and pressure oscillators, and the use of new high heat capacity materials to allow Gifford-McMahon and pulse tube refrigerators to reach temperatures below 4 K. A new cycle, known as the Boreas cycle, provides improved efficiency in small 4 K refrigerators and is also discussed. The status and future trends of these developments are presented.

INTRODUCTION

New applications of cryocoolers are appearing, and the requirements for old applications are often undergoing changes. These new and changing cryocooler requirements have been the impetus for the recent developments in cryocoolers. The lack of a suitable cryocooler to meet the requirements of a particular application has hampered the advancement of many applications. For example, superconductivity most likely would be in widespread use now if it were not for the problems associated with the cryocoolers needed to cool the superconductors. The main problems associated with cryocoolers are: unreliability, inefficiency, size, weight, vibration, and cost. The seriousness of each of these problems depends on the application. Within the last 10 years satellite applications for cooled infrared sensors have become much more important. Obviously these applications require a cryocooler with very high reliability (5 to 10 year lifetimes and no maintenance), high efficiency, small size, low weight, and low vibration. A significant amount of cryocooler development in the last 10 years has been focused on space applications. Because only a few cryocoolers are needed for this application at this time, cost has not been a serious problem. However, both cost and unreliability have been the major problems for most commercial applications. The application areas for cryocoolers are discussed first and then the recent advances made in the various types of cryocoolers are described.

APPLICATIONS AND REQUIREMENTS

Table 1 lists the major applications for cryocoolers that are currently in use or have some potential for large impacts. A more complete list is given by Radebaugh [1]. For many years the largest application

for cryocoolers (>100,000 manufactured to date in U.S. for this application) has been for use by the military in cooling infrared sensors to about 80 K for tactical applications. Refrigeration powers range from about 0.25 W to about 2 W. Stirling cryocoolers, used primarily for this application in the last ten years, have been able to meet the requirements, but their mean-time-to-failure (MTTF) of about 4000 hours is far short of the requirement for satellite and most commercial applications. Most of the new developments on Stirling cryocoolers has been in techniques to improve reliability. The rapid growth of research and development on pulse tube refrigerators in the last 5 years has been because of its potential for improved reliability and lower cost. The largest commercial application of cryocoolers has been for cryopumps (currently ~20,000/yr), which require a few watts of refrigeration at a temperature of about 15 K. Gifford-McMahon refrigerators have been used for this application. In the last year or two the vibration due to the moving displacer in the Gifford-McMahon refrigerator has become a problem in semiconductor fabrication as the circuit linewidths become narrower. A two-stage pulse tube refrigerator for such an application is now being studied at several laboratories.

- **Military**
    1. Infrared sensors for missile guidance
    2. Infrared sensors for surveillance (satellite based)
- **Police and Security**
    1. Infrared sensors for night vision and rescue
- **Environmental**
    1. Infrared sensors for atmospheric studies of ozone hole and greenhouse effects
    2. Infrared sensors for pollution monitoring
- **Commercial**
    1. Cryopumps for semiconductor fabrication
    2. High temperature superconductors for cellular-phone base stations
    3. Superconductors for voltage standards
    4. Semiconductors for high speed computers
    5. Infrared sensors for process monitoring
- **Medical**
    1. Cooling superconducting magnets for MRI systems
    2. SQUID magnetometers for heart and brain studies
    3. Liquefaction of oxygen for storage at hospitals and home use
    4. Cryogenic catheters and cryosurgery
- **Transportation**
    1. LNG for fleet vehicles
    2. Superconducting magnets in maglev trains
- **Energy**
    1. LNG for peak shaving
    2. Infrared sensors for thermal loss measurements
    3. Supercond. mag. energy storage for peak shaving and power conditioning
- **Agriculture and Biology**
    1. Storage of biological cells and specimens

Table 1. Applications of cryocoolers

TYPES OF CRYOCOOLERS

Figure 1 shows the schematics of the most common recuperative and regenerative cryocoolers. The recuperative types utilize a continuous flow of the refrigerant in one direction, analogous to a DC electrical system. As a result, the compressor and any expander must have inlet and outlet valves to control the flow direction, unless rotary or turbine compressors and expanders are used. The recuperative heat exchangers have two separate flow channels. In regenerative cycles the refrigerant undergoes an

oscillating flow and an oscillating pressure analogous to an AC electrical system. The compressor, or pressure oscillator, for the regenerative cycles needs no inlet or outlet valve. However, an oscillating pressure can be generated from a valved compressor by using another set of valves to switch between the high and low pressure sides of the compressor, as is done in the Gifford-McMahon refrigerator. The regenerator has only one flow channel, and the heat is stored for a half-cycle in the regenerator matrix, which must have a high heat capacity. Recent advances have occurred in all of these cryocooler types, which are discussed below. A new cryocooler type, known as the Boreas cryocooler [2], is a hybrid between a Gifford-McMahon and a Brayton cryocooler. Its use of recuperative heat exchange at low temperatures allows it to reach 4 K more efficiently than a pure regenerative cryocooler.

JOULE-THOMSON CRYOCOOLERS

Until recently closed-cycle Joule-Thomson cryocoolers have not been used in many applications because of low cycle efficiency and poor reliability of the compressor. Significant developments have occurred in the last few years regarding both of these problems, which have now brought this cycle from one of neglect to one of a serious contender for several applications. Plugging of the JT valve is still a persistent problem, but the use of self-regulating valves has eased the problem somewhat [3]. The Joule-Thomson cycle has an intrinsic inefficiency associated with the irreversible expansion in the JT valve or orifice. The efficiency of the expansion process becomes high only when the refrigerant is in the liquid or near-liquid state before passing through the JT valve, as in the vapor-compression cycle. The use of gas mixtures instead of pure nitrogen or argon gives rise to much greater enthalpy changes in the gas and a resulting improvement in the cycle efficiency, even at rather low pressures. An advantage of the Joule-Thomson cryocooler is the very low level of vibration because there are no moving parts or oscillating pressures in the cold head.

Mixed Refrigerants

The use of mixed refrigerants in the Joule-Thomson cycle has a fairly long and somewhat obscure history. Missimer [4] and Radebaugh [1] review some of this history and discuss some recent trends regarding mixed refrigerants. Joule-Thomson refrigerators using mixed refrigerants can be divided into two categories: those with phase separators and those without phase separators. In 1959 Kleemenko [5] used phase separators in a single flow stream which consisted of a gas mixture for the liquefaction of natural gas. The best performance was obtained with a mixture of 65 mol% methane, 20 mol% ethane, and 15 mol% normal butane. This single flow cycle had an efficiency better than that with two flow streams. The single stream cycle is now known as the mixed refrigerant cascade (MRC) cycle and is commonly used for the liquefaction of natural gas. It is sometimes referred to as the auto-refrigerated cascade

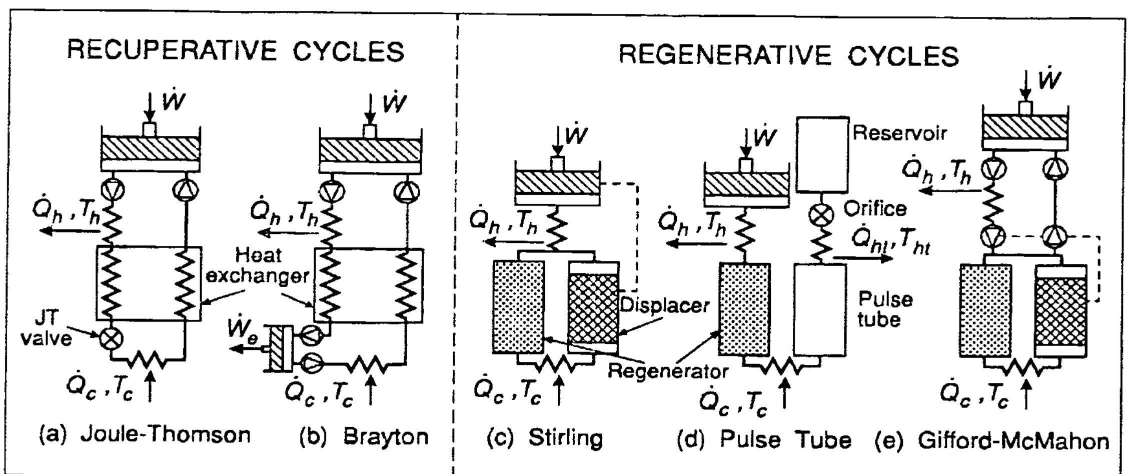

Figure 1. Schematics of five common cryocooler types.

(ARC) or the Kleemenko cycle. It can provide cooling at any number of stages through the use of phase separators and expansion valves. The phase separators allow only the liquid phase to be expanded at each stage, thus maintaining a high efficiency in the expansion process. The phase separators also are useful in reducing the amount of impurities, like water and oil, that are passed on to lower temperatures. In 1972 Missimer [6] used this same cycle with the addition of flow restrictions for added stability, but he replaced most of the hydrocarbons in the gas mixtures with CFCs and used oil lubricated air conditioning compressors for the cycle (known as the Polycold cycle) to provide about 300 W of refrigeration at 140 K. Recently the CFCs in this cycle have been replaced with non-CFCs. The Polycold cycle has been widely used for cold-trapping water vapor in vacuum systems. Little and Sapozhnikov [7] discuss progress on the use of the MRC or Kleemenko cycle for the cooling of CMOS multichip modules and other semiconductor electronic devices. They used a small domestic refrigerator unit with an appropriate multicomponent refrigerant and a single phase separator. They report success in maintaining temperatures down to 120 K for over 1000 hours before observing partial clogging due to oil carryover. Further work on gas mixtures and phase separators should lead to longer running times before clogging occurs.

In 1969 Fuderer and Andrija [8] were granted a German patent on a mixed refrigerant cycle in which there were no phase separators or intermediate expansion valves. The entire mixed refrigerant was cooled in the heat exchanger at which point it became 100% liquid before expanding through the orifice. This cycle is useful for applications where there is little need for refrigeration at intermediate temperatures. Temperatures down to 103 K were obtained by Fuderer and Andrija using a gas mixture of equimolar amounts of nitrogen, methane, ethane and propane and a 40:1 pressure ratio. They pointed out that because the mixture is in the two phase region in the heat exchanger, much greater heat transfer coefficients are possible compared with pure gas streams. In 1973 Alfeev *et al.* [9] from the Soviet Union used a similar mixture (30 mol% nitrogen, 30 mol% methane, 20 mol% ethane, 20 mol% propane) to achieve a temperature of 78 K using a 50:1 pressure ratio. The system efficiency was 10 to 12 times better than for pure nitrogen. Temperatures below 70 K were reached by adding neon, hydrogen or helium to the mixture. The application was for cooling infrared sensors to about 80 K, and there was no need for any significant cooling at intermediate temperatures. Even though methane, ethane, and propane have individual freezing points in the range of 85 to 90 K, some mixtures of these fluids have freezing points at least as low as 70 K [10]. Little [11] reviewed the Soviet work and discussed further advances with gas mixtures.

In 1994 Longsworth [12, 13] described a closed-cycle system using a commercially available oil-lubricated compressor (rolling-piston type) with a gas mixture of 36 mol% nitrogen, 20 mol% methane, 12 mol% ethylene, 20 mol% propane, and 12 mol% isobutane to obtain 1 W of cooling at 80 K and 10 W at 93 K with a high pressure of about 2 MPa. Compressor input power was about 350 to 400 W. At 80 K the maximum COP for the ideal JT cycle using the gas mixture described by Longsworth is about 50% of the Carnot COP. Further studies of optimum mixtures and of heat transfer in multicomponent, two-phase mixtures are needed to increase efficiency and to reduce the pressure needed to reach 80 K. When no phase separators are used, the solubility of oil in the refrigerant at low temperatures must remain fairly high. The use of hydrocarbons in the mixture helps to ensure a relatively high oil solubility. At present there is little published information on the solubility of oil in mixed refrigerants at low temperatures.

Sorption Compressors

In an effort to eliminate the only moving part in the Joule-Thomson cycle and improve the reliability of this cycle, the mechanical compressor has been replaced by sorption compressors in many recent studies. The sorption compressors also eliminate any vibration. Alternate heating and cooling of the sorption compressors cause a circulation of the refrigerant. This subject has been reviewed recently by Wade [14]. The only moving parts in the sorption compressors are check valves which operate once every few minutes to cause steady flow in one direction. Unfortunately, the sorption compressors have only been used with pure gases, which limits their efficiency unless many stages are used. Sorption of mixed gases may be possible, in principle, if the selective nature of sorption can be overcome. Chemisorption of hydrogen with hydrides (e.g., V and $LaNi_5$) and oxygen with praseodymium cerium oxide (PCO) can be done with the compressors operating at 300 K or even higher. Most other gases require the use of physisorption on microporous carbon. Low boiling point gases are difficult to adsorb at 300 K, so

physisorption compressors are generally limited to systems with cold temperatures above about 120 K. Alvarez *et al.* [15] discuss progress on an advanced sorption compressor system to reach 125 K using krypton with a xenon precooling stage at 170 K. They expect to achieve specific power inputs of 40 W/W at 125 K with a rejection temperature of 260 K. High efficiency in sorption coolers is only possible with the use of regenerative heating of the sorbent bed in which heat from one compressor module in a cool-down state is used to heat another compressor module in a warm-up state. Several modules are needed for high efficiency [16].

For temperatures down to about 65 K, Bard and Jones [17] proposed the use of a PCO compressor for an oxygen stage precooled with krypton and xenon stages using physisorption compressors. Specific powers of 50 W/W have been predicted for cooling to 65 K when using regenerative configurations [14]. Johnson and Jones [18] proposed the use of hydride beds pumping on solid hydrogen to reach temperatures down to 10 K in a periodic mode. A prototype space flight experiment (Brilliant Eyes Ten-Kelvin Sorption Cryocooler Experiment or BETSCE) of this concept has been developed and ground tested [19] and flown on the Space Shuttle in May 1996. During the ground tests a heat load of 100 mW was maintained at 9.5 K for over 20 minutes and the system could be recycled in under 5.5 hours. In this experiment Stirling cryocoolers were used to precool the hydrogen to about 65 K.

BRAYTON CRYOCOOLERS

The use of an expansion engine to carry out a reversible expansion of the gas as shown in Fig. 1(b) leads to higher efficiencies than are possible with Joule-Thomson cryocoolers. Most Brayton cryocoolers use turboexpanders with gas bearings to provide high reliability and very low vibration. The turboexpanders are very efficient in large sizes and are used in large liquefaction plants. For small cryocoolers the challenge is in fabricating the small turboexpanders and maintaining a high expansion efficiency. Recent advances in Brayton cryocoolers pertain to their miniaturization. Swift [20] discussed the latest developments in a single stage turbo-Brayton cryocooler designed to provide 5 W of cooling at 65 K. Turbomachines with gas bearings are used for both the compressor and the expander. The working fluid is neon with an inlet pressure of 0.11 MPa and a pressure ratio of 1.6. A specific power of 43 W/W was obtained with the engineering model Brayton cryocooler. With a cold temperature of 65 K and a reject temperature of 280 K, the Carnot efficiency is 7.7%. Though the turbomachines are very small (15 mm diameter compressor impeller and 3.2 mm diameter expander rotor), the slotted-disk heat exchanger is quite large (90 mm diameter by 533 mm in length). The total system mass is 11.9 kg, with the heat exchanger comprising 52% of the total mass. The high cost of miniature turbo-Brayton cryocoolers generally limits their use to space applications in which high reliability, high efficiency, and very low vibration are needed. An even smaller turbo-Brayton cryocooler is now being developed by McCormick *et al.* [21]. Their goal is to provide 2 W of refrigeration at about 65 K with less than 100 W of input power.

STIRLING CRYOCOOLERS

Most of the recent developments in Stirling cryocoolers have involved methods to improve reliability. Linear motor drives have replaced rotary motor drives in most applications because they eliminate many moving parts and reduce side forces between the piston and cylinder. Lifetimes of about 4000 hours are now routine for linear compressors with dry rubbing contact. Recent work by Pruitt [22] showed lifetimes in excess of 15,000 hours in linear compressors with rubbing contact. Wearout times of 3 years may be possible in some cases using improved materials for the rubbing contact. Such lifetimes may be sufficient for many commercial applications. The 5- to 10-year lifetimes needed for satellite applications as well as for some commercial applications can only be achieved when all rubbing contact is eliminated. Non-rubbing operation is achieved with piston devices by using flexure, gas, or magnetic bearings, or with diaphragm devices in which a flexing diaphragm causes compression and expansion of the working gas. Magnetic bearings are no longer being used in Stirling cryocoolers and will not be discussed further. Efficiencies for the conversion of electrical power to PV power have been as high as 85% in these linear compressors when operating at resonant frequencies.

Vibration caused by the reciprocating motion is reduced by using dual opposed pistons, a passive

balancer, or an active balancer. Vibration forces of about 1 N are typical in such cryocoolers that have input powers of the order of 100 W. Significant advances have been made in further vibration suppression in flexure-bearing cryocoolers by using active harmonic nulling with dual opposed pistons [23]. Axial vibration forces are often reduced to less than 0.1 N with this technique, though radial vibration forces remain unchanged at about 1 N [24].

Flexure Bearings

The most common technique for eliminating rubbing contact uses flexure bearings to support the piston and displacer inside their corresponding cylinders without any contact. A clearance gap of 10 to 20 μm provides the necessary flow impedance to serve as a dynamic seal. Figure 2a shows a simplified cross-section of a typical Stirling compressor with flexure bearings. A similar arrangement is used to support the displacer. In practice most compressors use two opposed pistons to eliminate most of the vibration. The flexure bearings provide a stiff support in the radial direction and act as a weak spring in the axial direction. The spiral geometry shown in the upper part of Fig. 2b is the geometry originally used by Davy of the University of Oxford in the early 1980s. Davy [25] reviews the development of these Oxford-style cryocoolers, which are now manufactured for space applications by several companies. The lower part of Fig. 2b shows a new flexure geometry proposed by Wong *et al.* [26] that offers a higher radial stiffness and less arm vibration than that of the spiral design.

Gas Bearings

Gas bearings have also been investigated for the elimination of rubbing contact between piston and cylinder. Work in France has focused on the use of hydrodynamic gas bearings, first in a Vuillemier cryocooler for space applications [27] and more recently in a two-stage Stirling cryocooler [28]. The linear motor used moving magnets and the rotation to drive the gas bearing was provided with a brushless motor. A rotational frequency of 5 to 10 Hz was used to provide the gas bearing effect in a 10 μm gap. A reciprocating frequency of about 35 Hz was used.

Hydrostatic gas bearings have been used in the U. S. for a Stirling cryocooler [29]. The gas flow for the bearing is driven by the oscillating pressure in the working space through a one-way valve (port

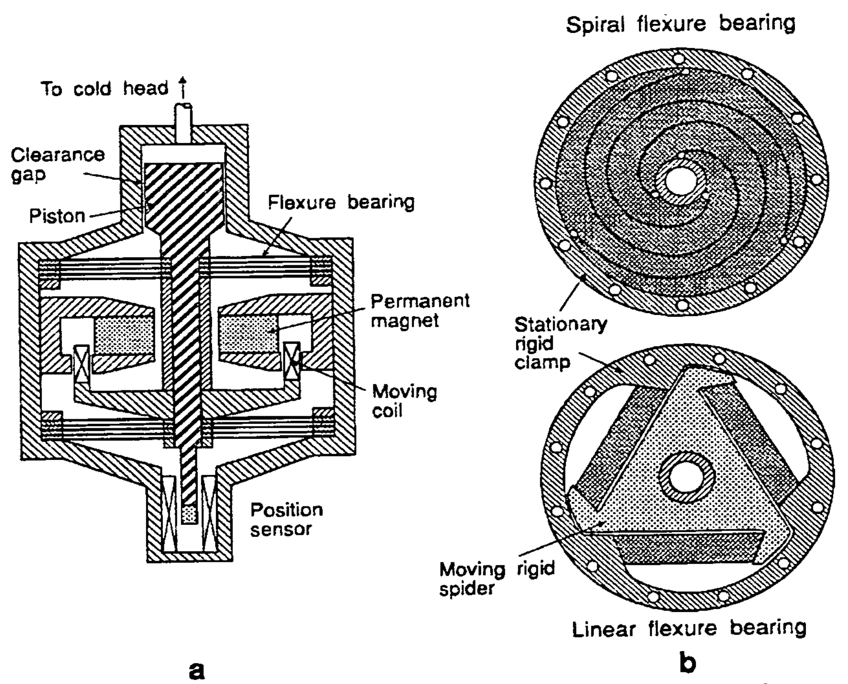

Figure 2. (a) Cross-section of the Oxford-style linear compressor showing the use of flexure bearings. (b) Two types of flexure bearings.

alignment between piston and cylinder could also be used to provide one-way flow) and flow restriction. An oscillating frequency of 60 Hz was used for this cooler.

Diaphragm Compressors and Expanders

A Ti-6Al-4V diaphragm with a flexible outer ring (similar to loudspeaker construction) has been used in both the compressor and expander assemblies of a single-stage 65 K Stirling cryocooler [30] and in a two-stage 30 K Stirling cryocooler [31]. This construction eliminates the need for clearance gaps and precise alignment. Such diaphragm devices tend to have short strokes and large diameters. The diaphragms were made double acting, which meant that the regenerator also functioned as a recuperative heat exchanger between the gas flowing in opposite directions in the two separate systems.

PULSE TUBE CRYOCOOLERS

The three main types of pulse tube refrigerators as discussed by Radebaugh *et al.* [32] are the basic, resonant (or thermoacoustic), and orifice. The orifice pulse tube refrigerator (OPTR) is the only one which achieves cryogenic temperatures (T<120 K) and is the only one considered in this review. Many versions of the OPTR now exist with various orifices or valves. Because the OPTR has no moving parts at the cold end, it has the following advantages over the Stirling cryocooler: (a) more reliable, (b) lower cost, (c) lower vibration, (d) lower EMI, (e) less sensitive to side loads, and (f) better launch survivability. However, until the last few years, its efficiency was much less than that of the Stirling refrigerator.

The OPTR operates with an oscillating pressure, which can be provided with a compressor like that of the Stirling refrigerator or with a Gifford-McMahon compressor and valves, with a sacrifice in efficiency. When using a Gifford-McMahon compressor the primary orifice and reservoir can be replaced by a set of switching valves connected to the compressor[33]. The OPTR operates on a cycle similar to the Stirling cycle except the proper phasing between mass flow and pressure is established by the passive orifice or switching valves instead of the moving displacer. The pulse tube provides a buffer volume of gas to maintain a temperature gradient between the hot and cold ends of the pulse tube. Any mixing within the buffer volume introduces a heat load on the cold end.

The analytical model developed at NIST in the late 1980s [34] for the OPTR solved the conservation of mass and energy equations with the assumption of simple harmonic pressure, mass flow, and temperature oscillations within the entire pulse tube refrigerator. That model also assumed adiabatic processes occurred within the pulse tube. Recent thermoacoustic theories [35-37] also use a harmonic approximation, but they include a linear approximation with higher harmonics and realistic heat transfer and viscous effects between the gas and tube walls. The non-adiabatic effects generally account for only a few percent loss in refrigeration power compared with the adiabatic model, unless the tube size is quite small. Other losses inherent in these models appear to result in a time-averaged enthalpy flow within the pulse tube that is about 70 to 80% of the ideal value [36, 37].

Improved Efficiencies

Because the expansion work is dissipated in the orifice instead of being recovered as in the Stirling cryocooler, the efficiency of an ideal OPTR compared with Carnot is given by

$$\eta_{ideal} = (T_h - T_c)/T_h. \tag{1}$$

Kittel [38] discusses this derivation. For T_c = 75 K and T_h = 300 K, η_{ideal} = 75%, whereas for T_c = 250 K, η_{ideal} = 17%. Thus, for cryogenic temperatures, practical inefficiencies dominate the intrinsic orifice inefficiency, but, for near-ambient temperatures, the intrinsic inefficiency severely limits the OPTR efficiency. The maximum possible COP for the OPTR is 1 when $T_c = T_h$.

At the time of the latest review of pulse tube refrigerators in 1990 [34], efficiencies of the OPTR were still several times less than that of Stirling refrigerators. Since 1990 several advances have been made in pulse tube refrigerators which have allowed them to achieve efficiencies nearly as high as that of Stirling refrigerators. A large OPTR constructed at NIST in 1991 produced 31.1 W of refrigeration at 80 K with a PV power input of 602 W at a rejection temperature of 316 K. The relative Carnot efficiency was 15.3% for PV work and 13% for electrical input power if the compressor were 85% efficient (typical of recent linear compressors). The average operating pressure was 2.5 MPa, and the

frequency was 4.5 Hz.

Improved efficiencies at higher operating frequencies were made possible by the introduction of the double inlet concept in 1990 by Zhu *et al.* [39]. They added a second orifice (often referred to as the secondary orifice or bypass orifice), as shown in Figure 3. (The intermediate orifice in this figure is discussed in the section on multiple stages.) With this orifice the gas flow needed to compress and expand the gas at the warm end of the pulse tube is taken directly from the compressor instead of passing through the regenerator and pulse tube. The reduced mass flow through the regenerator reduces the regenerator loss, particularly at high frequencies where the regenerator loss becomes quite large. A very efficient miniature pulse tube using the double inlet concept was reported by Chan *et al.* [40] in 1993. Their integral, inline pulse tube cryocooler operated at a frequency of 55 Hz and produced 0.53 W of cooling at 80 K with a compressor input power of 17.8 W and a reject temperature of 287 K (7.7% Carnot).

Recent research on high frequency OPTRs of sufficiently large size (larger than a few watts of gross cooling power) have shown that the inertia of the oscillating gas in a long tube (either the pulse tube [41], or the connecting tube between the pulse tube and the reservoir [42]), can be used to provide significant phase shifts between pressure and mass flow. The inertia produces a pressure component in phase with the acceleration of the gas. Because this pressure component leads the mass flow by 90°, it provides a beneficial phase shift between the pressure and the mass flow (pressure leading mass flow) like that produced by the secondary orifice but without the disadvantage of additional lost work and the potential for DC flow. The resistive component of the flow impedance in the long connecting tube (inertance tube in Fig. 3) can be made large enough, if desired, to eliminate the discrete primary orifice. In an optimally designed inertance tube the inertance (inductive) component of flow impedance is larger than that of the compliance (capacitance) component, so the pressure can be made to lead the mass flow at the warm end of the pulse tube. For gross cooling powers of a few hundred watts in a 45 Hz pulse tube, phase shifts in the inertance tube are large enough to eliminate the need for any secondary orifice.

Also, with large, high frequency OPTRs the mechanical driver can be replaced with a thermoacoustic driver (TAD) to produce cryogenic refrigeration with no moving parts[1,43]. A temperature of 90 K has been achieved with such a device, known as a TADOPTR.

A comparison of efficiencies for pulse tube refrigerators with Stirling refrigerators is shown in Fig. 4. The shaded band represents the efficiency range for most of the recent Stirling refrigerators. The circles in Fig. 4 show the efficiency of recent pulse tube cryocoolers which have achieved high efficiency. The highest power and highest efficiency pulse tube refrigerator is the NIST OPTR previously discussed. The lowest power OPTR is the miniature one by Chan *et al.* [40]. The two mid-size OPTRs are the same device, but with different input powers and different cold end temperatures [44]. Fig. 4 shows that the efficiencies of the latest pulse tube cryocoolers is almost as high as the best Stirling cryocoolers of a comparable size. Because of their high efficiency as well as many advantages over Stirling refrigerators, pulse tube refrigerators have been selected to cool the Atmospheric Infrared Sounder (AIRS), an important instrument used for the Earth Observing System (EOS) satellites in the study of the ozone hole and greenhouse effects.

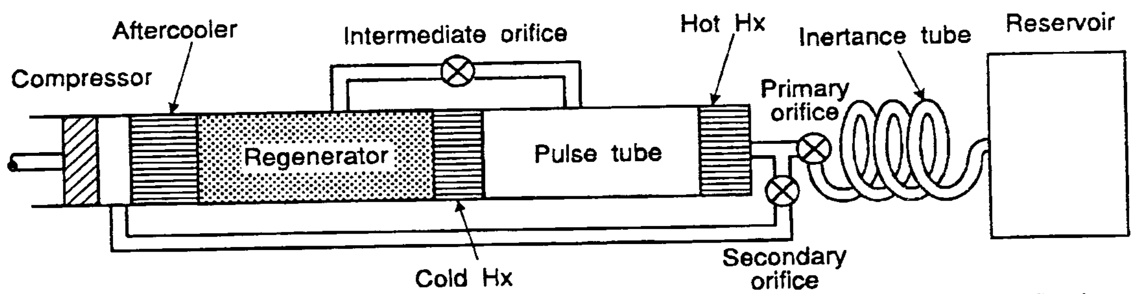

Figure 3. Schematic of the double inlet pulse tube refrigerator in which a secondary orifice is used. The multi-inlet concept for staging and the inertance tube are also shown.

Multiple Stages
For temperatures below about 80 K, two or more stages are normally used to maintain high efficiency. Two methods exist for the staging arrangement with pulse tube refrigerators. The first uses a separate pulse tube and regenerator for each stage with the warm end of each pulse tube at ambient temperature (parallel arrangement). If the pulse tube losses are too great for the second and lower stages, then their warm ends can be thermally anchored to the cold end of the next higher stage (series arrangement). These two arrangements, or a combination of the two, lead to the use of many pulse tubes, which may be cumbersome.

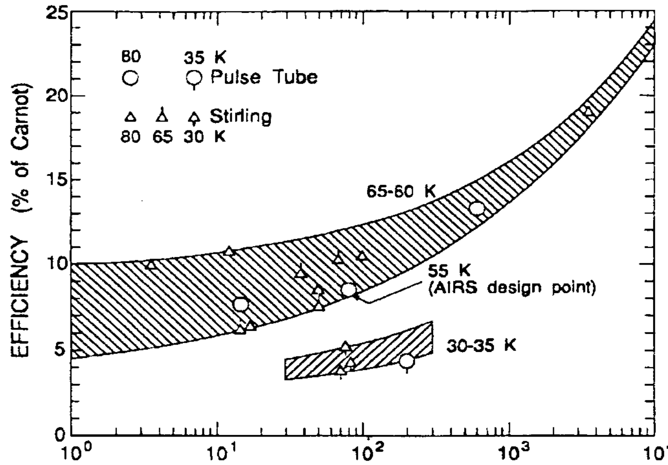

Figure 4. Efficiency of pulse tube refrigerators compared with that of Stirling refrigerators.

A new orifice arrangement, used by Zhou and Han in 1992 [45] and called a multi-inlet pulse tube, is in fact a new technique for staging. This arrangement, as shown in Fig. 3 and known as the multi-inlet arrangement, uses a middle or intermediate orifice to allow a portion of the gas to enter the pulse tube at an intermediate temperature, thereby producing refrigeration at that location. This staging arrangement maintains the simple geometry of a single pulse tube. Zhou and Han showed that the use of this intermediate orifice lowered the cold end temperature from 59 K to 33 K.

The lowest temperature achieved with a pulse tube refrigerator (PTR) was 3.6 K by Matsubara and Gao in 1994 [46] using a three-stage parallel arrangement. A low temperature of 2.07 K was just achieved by Thummes, *et al.* [47] with a two stage PTR precooled with liquid nitrogen. Some laboratories are now reaching 15 K with two stage PTRs [48]. These two stage PTRs have the potential of replacing the two stage Gifford-McMahon refrigerator in cryopump applications. Most of the two and three stage PTRs operate at about 2 Hz with switching valves and a Gifford-McMahon compressor. The reservoir volume is replaced with another switching valve connected to the compressor [33].

GIFFORD-McMAHON (GM) CRYOCOOLERS

Gifford-McMahon (GM) cryocoolers use oil-lubricated compressors made by the millions for the air conditioning industry. Even though they are modified for use with helium gas, their cost is quite low. The oil removal equipment is placed in the high pressure line ahead of the switching valve that generates the oscillating pressure. These cryocoolers are most commonly used in two-stage 15 K versions for cryopumps primarily for semiconductor fabrication. This is the largest commercial application of cryocoolers. The GM cryocooler is also used for cooling shields to 10 to 15 K in MRI systems to reduce the boiloff rate of liquid helium or for direct cooling of Nb_3Sn superconducting magnets to 10 K.

Most of the recent developments in GM cryocoolers have involved the use of high heat capacity regenerator materials to reach temperatures below 5 K without the aid of a Joule-Thomson stage. Rare-earth materials which undergo magnetic transitions in the 5-20 K range are generally used for these new regenerators. With Pb spheres replaced by Er_3Ni spheres in the second stage, Kuriyama, *et al.*[49] achieved a minimum temperature of 4.5 K with a two stage GM cryocooler in 1989. In the same year Nagao *et al.*[50] reach 3.3 K in a three stage GM cryocooler using crushed $Gd_xEr_{1-x}Rh$ powder. The development in Japan of the technique [51] to produce high quality spheres of many different rare-earth materials greatly aided the advancement of 4 K GM refrigerators. The lower porosity of 38% which spheres yield compared with about 50% for crushed powder decreases the regenerator loss at these low temperatures. Many different rare earth materials have been used in GM refrigerators since 1990. An excellent review of these materials and their applications is given by Hashimoto *et al.* [52]. Refrigeration

powers above 1 W at 4.2 K were first produced in 1993 by using two different materials in the second stage regenerator [53]. The highest efficiency, obtained with $ErNi_{0.9}Co_{0.1}$ spheres in the second stage, produced 1.6 W of refrigeration at 4.2 K with 7.1 kW of input power, or 1.6% of Carnot [54]. To date 2.2 W of refrigeration at 4.2 K is the most refrigeration produced by a GM cryocooler [55].

If a Joule-Thomson (JT) stage is added to the cold end of a GM cryocooler, much higher efficiencies are possible, but with higher cost and a sacrifice in reliability due to clogging of the JT valve. The highest efficiency for a combined GM/JT cryocooler at 4.5 K was reported by Fujimoto, *et al.*[56] where they produced 8.3 W at 4.4 K using 7.9 kW input, or 7.1% of Carnot.

BOREAS CRYOCOOLERS

The Boreas cryocooler was developed in the late 1980s specifically to fill the need for a small and efficient cryocooler for 4.2 K operation [2]. This cryocooler is a hybrid utilizing a three-stage GM cryocooler in which most of the fluid in the last stage, after expanding from 2 MPa to nearly atmospheric pressure, passes through a cold exhaust valve and returns through an outer concentric tube heat exchanger back to the compressor inlet. The returning cold gas precools the incoming steady flow of gas in the annular gap between the displacer and tube wall. Regenerative heat exchange in the gap also takes place simultaneously involving the oscillating component of gas flow. The gap regenerator is sufficient for precooling the gas to about 20 K, but for the last stage the recuperative heat exchanger dominates the overall heat transfer. The use of the recuperative heat exchanger removes the need for high heat capacity matrix materials like those used in a 4 K GM cryocooler. Because the expanded one atmosphere gas returns in a separate flow channel, this channel can be optimized for flow at one atmosphere to provide only a small pressure drop. The combination of using recuperative heat exchange at the low temperature end and the high pressure ratio (about 20:1) gives this cryocooler a higher efficiency than the GM cryocooler. For 1.1 W refrigeration at 4.2 K the input power is 2.9 kW, or 2.7% of Carnot. By contrast, the most efficient GM cryocooler has an efficiency of 1.6% of Carnot [54]. However, the Boreas cryocooler is not as efficient as the GM/JT cryocooler, which has an efficiency of 7.1% of Carnot [56]. The disadvantages of the Boreas cryocooler are the use of the moving cold exhaust valve in which there is little lifetime experience and the low refrigeration power at the upper stages caused by the use of a gap regenerator. Currently a smaller (0.25 W at 4.2 K) model is being developed which should have an input power of less than 1.5 kW and be air cooled. Potential application areas would be cooling of MRI magnets for the larger unit and the cooling of superconducting electronics for the smaller unit.

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0.5 W CLASS TWO-STAGE 4 K PULSE TUBE CRYOREFRIGERATOR

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ABSTRACT

Cryomech, Inc. has developed and commercialized a two-stage 4 K Pulse Tube Cryorefrigerator, the Model PT4-05. Because of the absence of moving parts and cold seals, the PT4-05 has extra lower vibrations, higher reliability and longer meantime between maintenance than the existing 4.2 K options. The PT4-05 is powered by the new Cryomech CP950 Compressor Package; consuming 4.7kW of input power, to provide 0.57 W at 4.2 K on the second stage with 18 W at 65 K on the first stage. The COP at 4.2 K is 1.2×10^{-4}. The PT4-05 reaches minimum temperature, 2.65 K, in 65 minutes. There has been no degradation in the performance of the PT4-05 presently under life test during more than 3,000 hours of operation and 90 cool-down and warm-up procedures. The meantime between maintenance is forecasted to be >20,000 hours.

INTRODUCTION

The unique characteristics of Pulse Tube Cryorefrigerator (PTR) will open new applications for cryorefrigerators due to the absence of displacer motion in the cold head. This characteristic promises greater reliability and lifetime while lowering the costs of operation and manufacturing when compared with GM and Stirling Cycle Cryocoolers.

To reach temperatures below 4 K in a multi-stage PTR , special consideration must be given to the configuration of the stages and to the composition or the magnetic regenerative materials. Matsubara and Gao[2] in Japan first obtained a temperature of 3.6 K in 1994 with a 3-stage pulse tube cryorefrigerator. The net cooling power of their refrigerator at 4.2 K of 30 mW would be too low for the most applications. More recently, Prof. Heiden's group (which included the first author) at the University of Giessen developed a two-stage 4 K pulse tube, which obtained a lowest temperature of 2.2 K as well as several hundred milliwatts at 4.2 K in 1997. The group applied this PTR successfully to both liquefy ^{4}He with a rate of 3 liters/day[4] and to conductively cool a 2.8 Tesla superconducting magnet to below 4 K[5].

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In this paper, we are reporting on the two-stage 4 K PTR that has been developed at Cryomech. The PT4-05 has achieved the highest COP,1.2×10^{-4} to date 4.2 K of any PTR.

DESIGN OF PT4-05

The two-stage 4 K pulse tube cryorefrigerator includes a pulse tube cold head and a helium compressor package. A new compressor, Cryomech Model CP950, was designed for the pulse tube with regard to a long meantime between maintenance (MTBM). The compressor package supplies the cold head with pressurized helium through flexible metal hoses. A rotary valve in the cold head, similar to that in GM cryorefrigerator, directs the helium gas in and out of the pulse tube system.

1. PT4-05 Cold head

Figure 1 shows a photograph of the PT4-05 cold head. There is a schematic of the pulse tube cryorefrigerator displayed in Figure 2. The two-stage pulse tube cryorefrigerator employs the double-inlet configuration. Two reservoirs, four orifices and a rotary valve control the helium flow in the PTR. They are integrated inside the motor mount assembly at the warm end, which also performs as the room temperature heat sink for the refrigerator. The 1st stage cold plate, which provides cooling power between 40 K to 75 K, thermally anchors the 1st stage cold heat exchanger, the cold end of the 1st stage regenerator and 2nd stage pulse tube. The 2nd stage heat exchanger can provide cooling power below 4 K.

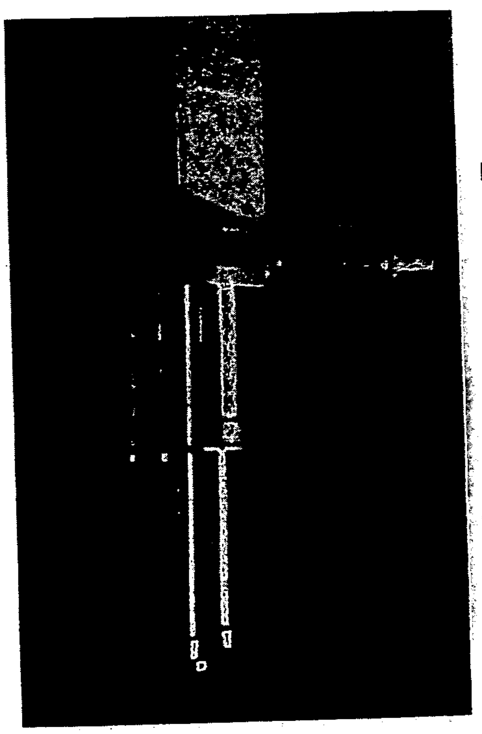

Figure 1. Photograph of the PT4-05 cold head

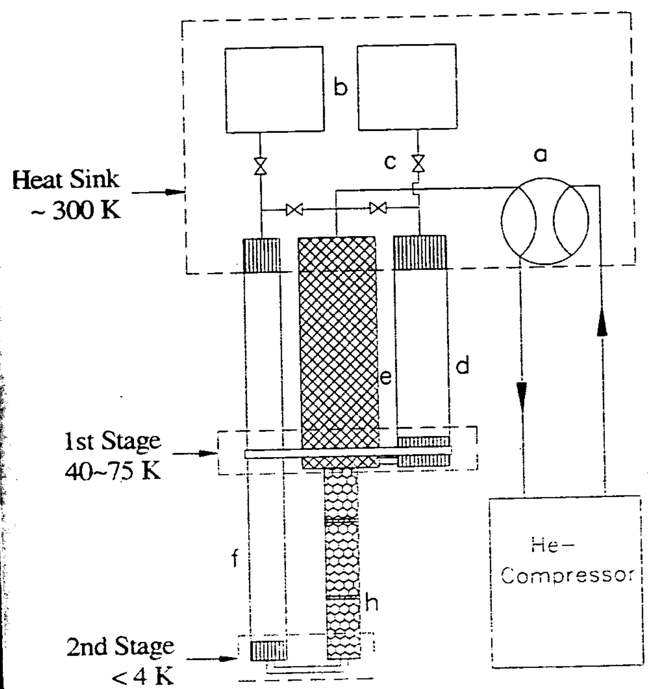

Figure 2. Schematic of PT4-05. a: rotary valve; b: 1st and 2nd stage reservoirs; c: orifices; d: 1st stage pulse tube; e: 1st stage regenerator; h: 2nd stage regenerator; f: 2nd stage pulse tube.

Table 1. Key Features & Benefits of PT4-05

Feature	Benefit
Cold head	
Pulse tube with double-inlet configuration	Extra low vibration, greater reliability and lifetime
Integrated rotary valve and pulse tube	Very compact and reliable, easy coupling
DC stepper motor driven valve	Very quiet operation and low electronic noise
Compressor package	
Helium scroll compressor	Highly reliable, efficient, and low in noise and vibration level
Highly efficient absorber design	MTBM > 20,000 hours

2. New Compressor Package, CP950

MTBM for a normal helium compressor packages is around 10,000 hours, which is too short when compared to that of the pulse tube cold head. Therefore, we developed a new compressor package, the CP950, which employs a helium scroll compressor and an highly efficient absorber with expected lifetime > 20,000 hours.

The key features and benefits of the PT4-05 are listed in Table 1. The DC stepper motor, which turns the rotary valve, is very quiet, and reduces electrical noise. Outside testing of vibration levels have not been able to quantify the level of vibration; because its vibration spectra is below that of the testing facilities environment; which was around 1.0×10^{-2} g. The meantime between maintenance of the PT4-05 is expected to be > 20,000 hours.

PERFORMANCE OF PT4-05

Figure 3 shows the cool-down characteristics of PT4-05. It takes 65 min for the 2nd stage to reach the minimum no load temperature of 2.65 K, and 120 min for the 1st stage to reach the no load temperature of 39 K. The temperature of the pulse tube at 4 K is extremely stable, fluctuating less than 0.2 K which is much smaller than the 1-0.5 K in a typical GM[6]. When the PT4-05 operates with no load, at the minimum temperatures, the input power of compressor is 4.3 kW.

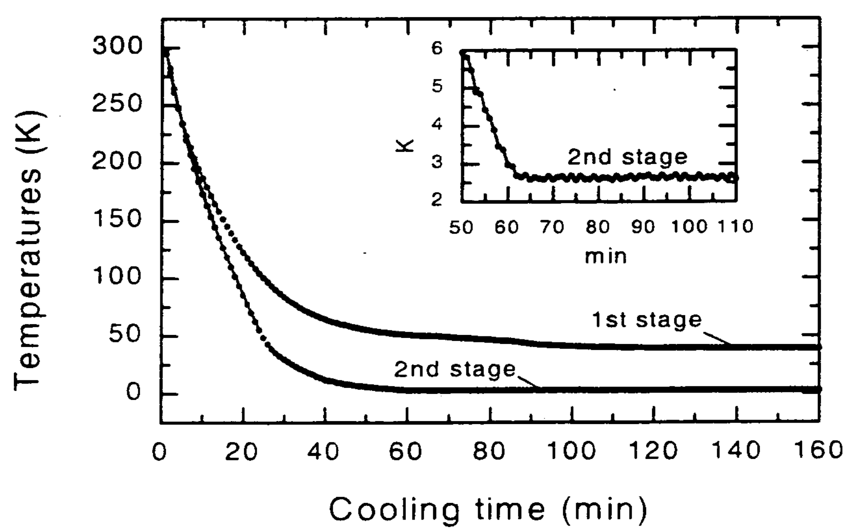

Figure 3. Cool-down performance of PT4-05

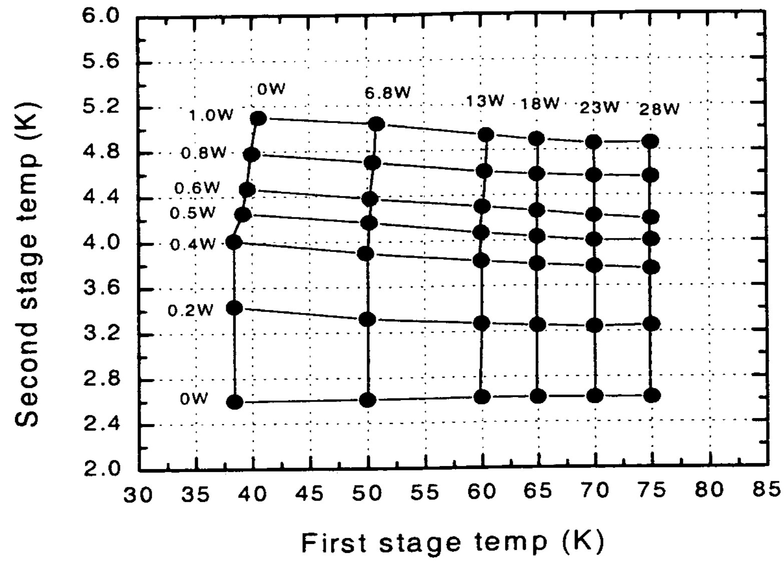

Figure 4. Typical cooling load map of PT4-05

A cooling load map of PT4-05 is given in figure 4. It provides 0.57 W at 4.2 K on the 2[nd] stage simultaneously with 18 W at 65 K on the 1[st] stage, while consuming 4.7 kW of input power. When the 1[st] stage provides 28 W at 75 K the 2[nd] stage can provide 0.61 W at 4.2 K. The increase in cooling power at 4.2 K upon increasing the 1[st] stage temperature (heat load) is caused by the increased pressure differential now available due to the higher temperature at the 1[st] stage. This also increases the compressor input power.

Figure 5 shows the cooling power curve of the 2[nd] stage, while the 1[st] stage has 18 W at 65 K. At higher temperatures the PT4-05 has a 2[nd] stage cooling power of 1.0 W at 4.8 K and 2.0 W at 7.0 K. In Table 2, the performance of PT4-05 is compared to the first two-stage 4 K pulse tube developed at University of Giessen[7]. The COP of the PT4-05 at 4.2 K is 1.2×10^{-4}, which has doubled.

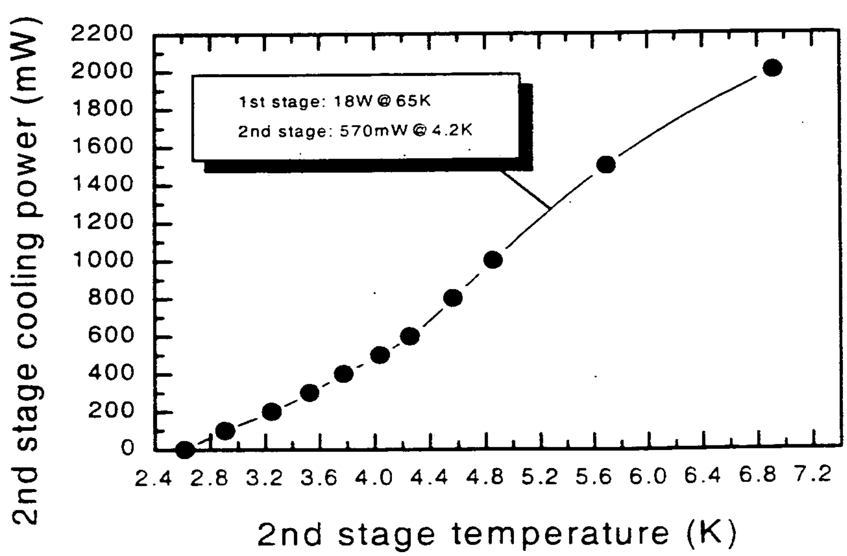

Figure 5. 2[nd] stage cooling power vs. temperature

Table 2. Performance comparison between two 4 K pulse tube cryorefrigerators

	2-stage PT at Uni. of Giessen[7]	*Cryomech PT4-05*
Input power	6.3 kW	4.7 kW
2nd-stage cooling power	0.42W @ 4.2K simultaneously 20W @ 67K	0.57W @ 4.2K simultaneously 18W @ 65K
COP @ 4.2K	0.6×10^{-4}	1.2×10^{-4}

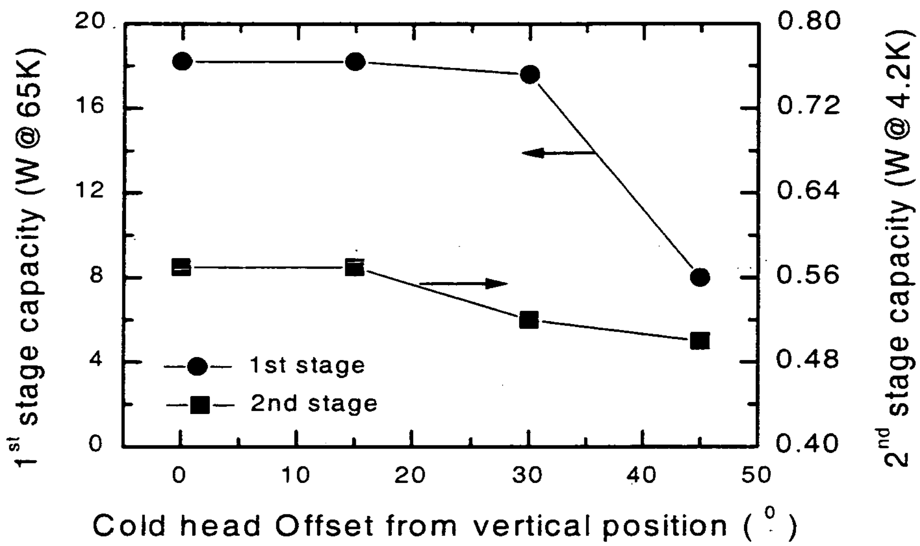

Figure 6. Orientation effects on the cooling performances

The PTR has been tested for orientation affects. It is well documented that tilting a PTR will degrade its performance. This is caused by the natural convection of the helium, and is more prevalent in low frequency pulse tubes. Figure 6 shows the effects of orientation on the performance of the PT4-05. The vertical axes show the cooling powers on the 1st temperature at 65 K and the 2nd stage at 4.2 K. There was no degradation in performance on both stages if the pulse tube cold was tilted 15^0 from the vertical position. For a tilt of 30^0, the cooling power on the 2nd stage drops down to 520 mW from 570mW. For a tilt of 45^0, the cooling power on the 2nd stage decreases to 500 mW, while the 1st stage decreases to 8 W from the 17 W at vertical. Therefore it is suggested that the pulse tube cold head operates as close to vertical as possible. But slight variations from vertical up to 30° will not degrade the performance greatly.

The pulse tube has been cooled down and warmed up more than 90 times, has operated for more than 3,000, and has continuously operated for more than 2,000 hours. Figure 7 shows the temperature stability on the 1st stage and 2nd stage during the continuously operation. There has been no performance degradation on the 2nd stage during the test. The 1st stage temperature slightly increased from 39 K to 40 K because of the variation of environment temperature. The life test of PT4-05 reported on in this paper, started in the past winter and ended this summer.

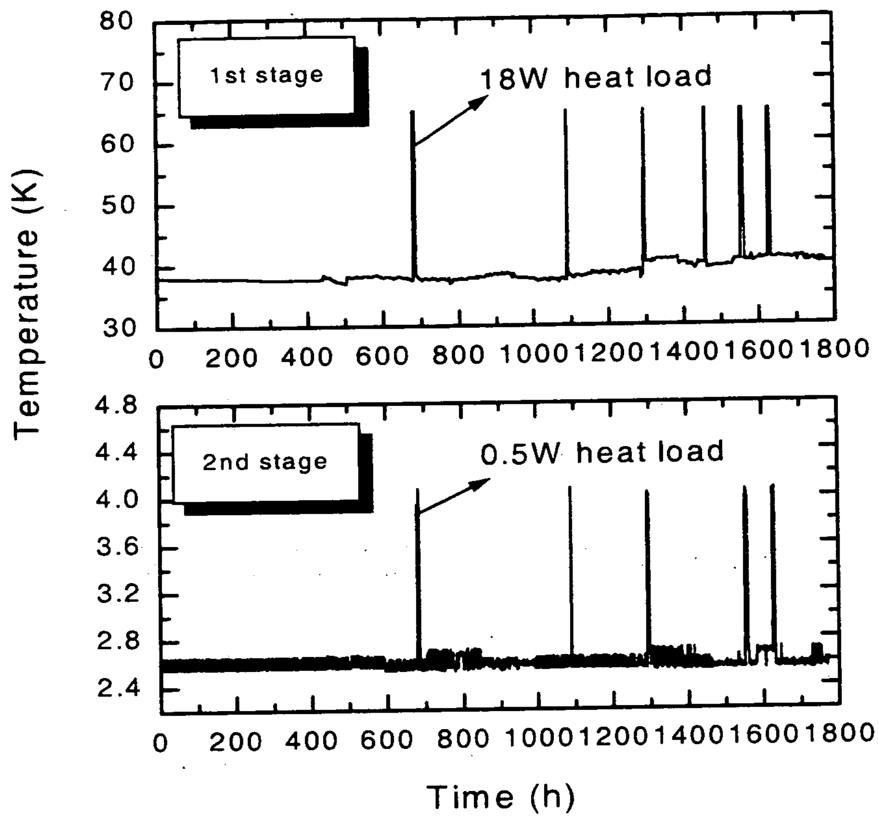

Figure 7. Temperature stability of PT4-05 during long term operation

CONCLUSION

A two-stage 4 K pulse tube cryorefrigerator has been was successfully developed and commercialized at Cryomech, Inc. The pulse tube can provide 0.57 W at 4.2 K on the 2nd stage and simultaneously 18 W at 65 K on the 1st stage, for 4.7 kW of the compressor input power.

The pulse tube cold head has the features of extra low vibration as well as 20,000 hours MTBM, and will open up attractive applications in cooling detectors, superconducting and cryoelectronic devices.

The two-stage pulse tube with cooling power of 1 W at 4.2 K is being developed at Cryomech now. A cooling power of 0.72 W at 4.2 K has already been obtained by using a larger compressor.

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